

Article

Influencing Factors of Failure Modes in High-Speed Permanent Magnet Motor Rotors Under Axial Constraint

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Abstract

High-speed permanent magnet machines frequently employ axially constrained solid rotors. These rotors undergo complex failure due to the combined action of centrifugal loading and interference-fit contact pressure. Existing analytical models often overlook this interaction, which can lead to overestimated reliability. To address this limitation, a three-dimensional axisymmetric elastic–static model is developed. This model quantifies how rotational speed and interference fit jointly govern failure behaviour. A systematic computational framework is established to evaluate three primary failure modes: interface debonding, permanent magnet fracture, and retaining sleeve yielding. The framework explicitly accounts for axial constraint effects. Finite element simulations and prototype testing on a 3 kW, 200,000 r/min motor validate the model, confirming its accuracy in predicting failure limits. The stress distributions and dynamic responses under axial constraint are systematically analysed and compared with numerical and experimental results. Consequently, the proposed approach provides a reliable foundation for designing high-speed rotors under axial constraints.

Keywords: permanent magnet motors; rotor dynamics; mechanical strength; failure analysis; high-speed applications

1. Introduction

High-speed permanent magnet machines (HSPMMs) deliver high power density and high operating efficiency, making them suitable for a range of applications. Examples include air compressors, high-speed pumps, flywheel energy-storage systems, and low-altitude unmanned aerial vehicle propulsion [1–4]. These advantages arise from their compact design and high performance. However, high-speed operation raises significant concerns regarding rotor structural integrity. In service, the rotor sustains substantial centrifugal loads which increase quadratically with speed. The rotor is further subjected to coupled electromagnetic, thermal, and mechanical stresses. Under these demanding conditions, mechanical failure can occur. Such failure risks both system safety and operational lifespan.

High-speed permanent magnet machines (HSPMMs) frequently use surface-mounted permanent magnet (SPM) rotors. This design is popular because of its simplicity and low cost. However, SPM rotors present reliability concerns. They are prone to three main failure



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modes: interfacial detachment, permanent magnet fracture, and yielding of the retaining sleeve [5–8].

Interfacial detachment is a critical failure mode. It primarily originates from two sources: centrifugal forces and thermal expansion. These factors jointly reduce the interference-fit pressure between rotor components. This reduction may progress to complete interfacial separation, ultimately causing system failure. Magnet fracture results from the low tensile strength of sintered permanent magnets. A fracture occurs when the total tensile stress exceeds the material's ultimate strength. This stress arises from centrifugal, assembly, and thermal loads. Sleeve yielding develops under multiaxial stress from centrifugal, interference, and thermal effects. According to the von Mises criterion, plastic deformation occurs when these stresses surpass the material's yield strength.

In practical rotor assemblies, axial constraints are often present, such as end plates and lock nuts. These constraints increase design complexity. These constraints limit free thermal expansion and generate additional axial stresses. Consequently, the distribution of radial and circumferential stresses within the rotor is altered. Ignoring these three-dimensional constraint effects can lead to significant errors in predicting mechanical stress and failure limits. This is particularly true under conditions of repeated heating and cooling.

Recent research has made considerable progress in predicting stress and preventing failure in high-speed permanent magnet rotors. Improved models and new materials have driven these advances. Researchers now increasingly use multiphysics-coupled models [9–11], data-driven approaches [12,13], and material-specific methods [14,15]. These methodologies collectively contribute to creating more robust designs. Furthermore, electromagnetic aspects such as slot configuration and material selection are pivotal for achieving high motor efficiency. Their thermal outcomes must be reconciled with mechanical stress distributions, as highlighted in recent studies [16].

For example, Wang et al. [17] developed a detailed model for a rotor operating at 28,000 r/min. Their approach combined advanced material laws with efficient computational methods, enabling the accurate prediction of stress and the determination of safe operating speeds. In the field of fracture mechanics, Ben Said et al. [18] employed a refined numerical approach to model complex crack growth, identifying the stress levels that triggered different failure modes. For electric vehicle applications, Yu et al. [19] developed and experimentally validated a comprehensive multiphysics model of an 80,000 r/min rotor. Their work coupled electromagnetic, thermal, and mechanical effects to explain failure mechanisms and guide design improvements. Additionally, Li et al. [20] proposed a fast analytical model to estimate peak stress in interior permanent magnet motors. This model incorporated detailed thermal analysis, thereby improving prediction accuracy during the motor design stage.

Nevertheless, finite element analysis is still the main tool in current research on high-speed permanent magnet rotors. It is used for both detailed calculation and general pattern recognition. In contrast, comprehensive analytical modelling frameworks are not yet well established. Recent work has aimed to close this gap by improving analytical methods for stress prediction. For example, Zeng et al. [21] proposed an analytical model. Their model includes pre-stress, centrifugal forces, and thermal effects at the same time. This model provides a practical way to evaluate stresses in anisotropic composite sleeves. Following this, Yang et al. [22] introduced the concept of virtual interference. They also formulated a unified stress field model based on two-dimensional polar coordinates. Their study systematically explains how different interference-fit methods change stress distributions inside the rotor.

Ge et al. [23] improved multiphysics models for cryogenic applications. Their models include nonlinear material properties. This approach helps explain how extreme cold

interacts with mechanical stress. The same research group also combined response surface methodology with multi-objective optimisation. This integration has produced a design framework that can enhance both structural strength and weight reduction simultaneously. In a related study, Xu et al. [24] refined analytical computation methods. They achieved this by using equivalent thermal resistance networks. Recently, Liu et al. [25] introduced physics-informed neural networks for calculating rotor stress. This development creates new possibilities for real-time system monitoring.

Despite progress in rotor strength assessment, current research still faces three main limitations. (i) Most existing analytical models are based on two-dimensional plane-stress assumptions. These simplified models omit significant three-dimensional effects. They do not adequately account for the significant stresses caused by axial constraints. (ii) A systematic analytical understanding is still missing. Such understanding would explain the coupling mechanisms and parameter sensitivities. These factors control different rotor failure modes, including sleeve yielding, magnet fracture, and interfacial debonding. (iii) These limitations make it difficult to determine the safe operating range for high-speed rotors accurately. They also reduce the reliability of the design methods that follow.

To address these gaps, this paper proposes a new three-dimensional analytical framework. This framework explicitly incorporates axial-end constraints and evaluates their interaction with competing failure modes. The study uses a 3 kW, 200,000 r/min solid permanent magnet machine as the reference design for analysis.

It is essential to note that this study primarily focuses on the mechanical aspects of rotor failure under axial constraint. It investigates the coupled effects of interference fit, centrifugal loading, and thermal loading. Electromagnetic forces are recognized as essential contributors to rotor stress in operating motors. Including them in the current analytical framework would significantly increase model complexity. Therefore, electromagnetic effects are beyond the scope of this initial investigation. The proposed model provides a foundation for understanding constraint-induced stress redistribution. This foundation can be extended in future work to incorporate electromagnetic–mechanical coupling effects.

The principal contributions are summarised as follows:

- A three-dimensional static analytical model is developed for axially constrained rotors. Grounded in axisymmetric elasticity theory, the model establishes closed-form governing equations for the rotor stress field. These equations facilitate the quantitative assessment of constraint effects.
- Computational frameworks are established to evaluate three primary failure modes. These failure modes include interface debonding, permanent magnet fracture, and yielding of the retaining sleeve. These frameworks integrate dynamic interference-fit models. They also incorporate displacement coupling and stress modification correction factors. This integration enables efficient quantitative assessment under combined centrifugal and interference loading.
- The synergistic effects of rotational speed and interference fit are systematically investigated. The model elucidates how these parameters collectively influence the three failure criteria. It establishes a theoretical foundation for optimising rotor design.
- Comprehensive validation is conducted through finite element simulations and prototype testing. The analytical model and computational methodology are verified via comparative finite element analysis. Additionally, shaft orbital response measurements provide indirect corroboration of the model's predictive capability. These measurements are obtained from a custom-fabricated high-speed motor prototype. They validate the model's prediction of stress amplification under axial constraint.

2. Materials and Methods

This section describes the configuration and material properties of the solid-rotor motor being studied. The theoretical framework is then introduced. It focuses on a static model for rotors under axial constraint. Finally, the primary factors affecting rotor strength failure modes are examined. This examination leads to the development of evaluation models for key performance parameters.

2.1. Motor Configuration and Material Parameters

The high-speed permanent magnet motor in this study has a concentric structure. It includes, from outside to inside, the housing, stator core, air gap, and rotor assembly (see Figure 1). This motor operates based on the radial-flux principle. It uses a solid cylindrical permanent magnet in a surface-mounted configuration. The rotor assembly primarily consists of a retaining sleeve and permanent magnets. An interference fit connects these components to ensure structural stability at high speeds. Table 1 lists the principal dimensions and structural parameters of the motor.

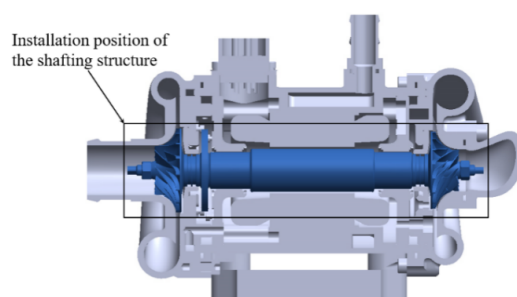


Figure 1. Configuration of the high-speed permanent magnet motor.

Table 1. Key dimensions and fundamental parameters of the permanent magnet motor.

Parameter	Value	Parameter	Value
Rated power (kW)	3	Rated speed (r/min)	200,000
Stator outer diameter (mm)	53	Stator inner diameter (mm)	26
Number of stator slots	12	Air-gap length (mm)	3
Sleeve outer diameter (mm)	20	Magnet outer diameter (mm)	17
Sleeve length (mm)	55	Magnet length/mm	37

The rotor assembly includes an impeller, a shaft, a thrust disk, permanent magnets, a retaining sleeve, and a lock nut (see Figure 2). A central shaft subassembly serves as the rotor core. In this core, the retaining sleeve, permanent magnets, and shaft end are connected using interference fits. This connection maintains structural integrity and ensures reliable operation. The solid permanent magnets are held in place axially. They are constrained between the thrust disk and end plugs to prevent axial movement. Table 2 provides the materials and properties of all rotor components.

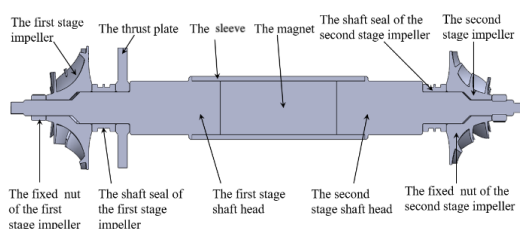


Figure 2. Schematic of the rotor assembly.

Table 2. Material properties of rotor components.

Component	Material (Grade/Common Name)	Density (g·cm ^{−3})	Poisson's Ratio	Elastic Modulus (GPa)	Tensile Yield (MPa)	Compressive Yield (MPa)
Impeller	Aluminum Alloy 2618A (Forged) [26]	2.76	0.33	74.5	360	–
Shaft End	Stainless Steel AISI 4169 (Martensitic) [27]	8.24	0.30	204	1030	–
Thrust Plate	Stainless Steel AISI 4169 (Martensitic) [27]	8.24	0.30	204	1030	–
Retaining Sleeve	Stainless Steel AISI 4169 (Martensitic) [27]	8.24	0.30	204	1030	–
Permanent Magnet	Samarium-Cobalt Sm ₂ Co ₁₇ (Sintered, Grade: YX-30H) [28]	8.4	0.28	117	600	85

2.2. Static Mechanical Model of an Axially Constrained Rotor

High-speed solid permanent magnet motors use integrated rotors. In these rotors, retaining sleeves and permanent magnets are joined by interference fits. In practice, mechanical constraints at the rotor's axial ends result in strong axial confinement. These constraints are typically imposed by end plates or retaining rings. This confinement limits free thermal expansion along the axis during operation. As a result, axial stresses are induced, and the distributions of radial and circumferential stresses are significantly altered. Based on three-dimensional axisymmetric elasticity theory, governing equations for the stress field in the solid rotor are formulated. These equations explicitly include the effects of axial constraints. Corresponding analytical solutions for the stress and displacement fields are then systematically derived.

The rotor is treated as an isotropic material in the cylindrical coordinate system (r, θ, z) . Equation (1) provides the strain–displacement relationships.

$$\varepsilon_r = \frac{\partial u}{\partial r}, \quad \varepsilon_\theta = \frac{u}{r}, \quad \varepsilon_z = \frac{\partial w}{\partial z}, \quad \gamma_{rz} = \frac{\partial u}{\partial z} + \frac{\partial w}{\partial r} \quad (1)$$

Here, u and w represent the radial and axial displacements, respectively; r and z denote the corresponding radial and axial coordinates. Within this coordinate system, the normal strain components are ε_r , ε_θ , and ε_z , and γ_{rz} is the shear strain component.

The constitutive relationship follows the generalised Hooke law including thermal effects, as given in Equation (2).

$$\begin{aligned} \varepsilon_r &= \frac{1}{E_r} [\sigma_r - \nu_{r\theta}\sigma_\theta - \nu_{rz}\sigma_z] + \alpha_r \Delta T, \\ \varepsilon_\theta &= \frac{1}{E_\theta} [\sigma_r - \nu_{\theta r}\sigma_r - \nu_{\theta z}\sigma_z] + \alpha_\theta \Delta T, \\ \varepsilon_z &= \frac{1}{E_z} [\sigma_z - \nu_{zr}\sigma_r - \nu_{z\theta}\sigma_\theta] + \alpha_z \Delta T \end{aligned} \quad (2)$$

Here, E_r , E_θ , and E_z denote the elastic moduli in the radial, circumferential, and axial directions, respectively. ν_{ij} ($i, j = r, \theta, z$) represents the Poisson's ratios. It is defined as the negative ratio of transverse strain in the j -direction to axial strain in the i -direction when stressed in the i -direction. α_r , α_θ , and α_z are the coefficients of thermal expansion in the corresponding directions. σ_r , σ_θ , and σ_z refer to the standard stress components in the cylindrical coordinate system. ΔT represents the temperature rise relative to the assembly reference state.

The stress distribution in a solid rotor satisfies the equilibrium differential equation for axisymmetric rotating bodies [29], given by Equation (3).

$$\frac{\partial \sigma_r}{\partial r} + \frac{\partial \tau_{rz}}{\partial z} + \frac{\sigma_r - \sigma_\theta}{r} + \rho \omega^2 r = 0 \quad (3)$$

Here, τ_{rz} denotes the shear stress in the r - z plane, ρ represents the material density, and ω is the rotational angular velocity. Under the assumptions of uniform axial stress ($\partial \sigma_{ij} / \partial z = 0$) and negligible shear stress gradients ($\partial \tau_{rz} / \partial z = 0$), Equation (3) simplifies to Equation (4).

$$\frac{\partial \sigma_r}{\partial r} + \frac{\sigma_r - \sigma_\theta}{r} + \rho \omega^2 r = 0 \quad (4)$$

The key difference between the proposed model and conventional models is the axial constraint. This constraint prevents overall axial movement during operation. Consequently, the global axial strain is zero ($\varepsilon_z = 0$). Substituting σ_z into Equation (2) gives modified expressions for radial strain ε_r and circumferential strain ε_θ . These modified expressions are shown in Equation (5).

$$\sigma_z = \nu_{zr} \sigma_r + \nu_{z\theta} \sigma_\theta - E_z \alpha_z \Delta T \quad (5)$$

Here, under axial constraint, axial stress σ_z depends linearly on radial stress σ_r , circumferential stress σ_θ , and thermal load ΔT . Substituting Equation (5) for σ_z into Equation (2) yields modified expressions for radial strain ε_r and circumferential strain ε_θ . These modified expressions are given in Equation (6).

$$\begin{aligned} \varepsilon_r &= a_{11} \sigma_r + a_{12} \sigma_\theta + \beta_r \Delta T, \\ \varepsilon_\theta &= a_{21} \sigma_r + a_{22} \sigma_\theta + \beta_\theta \Delta T \end{aligned} \quad (6)$$

Here, the coefficients a_{ij} and β_k ($i, j = 1, 2; k = r, \theta$) are material constants. The coefficients characterize the coupling effects caused by axial constraints. They characterise the coupling effects caused by axial constraints. Under axial constraint, the governing equation for radial displacement $u(r)$ is obtained by substituting Equation (6) into Equation (3). It gives Equation (7).

$$\frac{d^2 u}{dr^2} + \frac{1}{r} \frac{du}{dr} - \frac{\lambda}{r^2} u = f(r) \quad (7)$$

Here, λ is a dimensionless constant derived from the coefficients in Equation (6). It quantifies the influence of material anisotropy and Poisson's coupling under axial constraint within the displacement equation. The non-homogeneous term $f(r)$ contains the rotational body force $\rho \omega^2 r$ and the gradient of thermal strain $d(\alpha \Delta T) / dr$. This term drives the particular solution of the displacement field in Equation (8).

$$u(r) = C_1 r^m + C_2 r^n + u_p(r) \quad (8)$$

Here, C_1 and C_2 are integration constants determined by the boundary conditions; m and n denote exponents related to the parameter λ ; and $u_p(r)$ represents the particular solution. This solution represents displacement components due to centrifugal forces and thermal loads.

For an interference-fit rotor assembly with an inner permanent magnet and an outer retaining sleeve, the boundary conditions are defined as follows:

- At the permanent magnet centre ($r = 0$), the displacement remains finite: $u_{pm}(0) = 0$;

- On the outer surface of the retaining sleeve ($r = R_o$), the radial stress boundary condition requires: $\sigma_{r,sleeve}(R_o) = 0$;
- At the interface between the sleeve and the permanent magnet ($r = R_i$), displacement compatibility and stress continuity are enforced according to Equation (9).

$$\begin{aligned} u_{sleeve}(R_i) - u_{pm}(R_i) &= \delta_{eff}, \\ \sigma_{r,sleeve}(R_i) &= \sigma_{r,pm}(R_i) = -P_{contact} \end{aligned} \quad (9)$$

Here, δ_{eff} denotes the effective dynamic interference. It combines the initial assembly interference with the effects of centrifugal force and thermal expansion. $P_{contact}$ represents the contact pressure at the interface.

2.3. Assessment of Rotor Strength Failure Modes

In practical assemblies, axial constraints are not perfectly rigid. It is due to the flexibility of end plates, bolts, and other components. This flexibility permits a limited amount of axial elongation under operational loads. The present model treats the axial constraint as a zero-global-axial-strain condition for deriving the stress field. This approach yields a conservative estimate of axial stress. It also captures the coupling between axial stress and radial and circumferential stresses. The sensitivity of interface debonding and sleeve stress to changes in axial constraint stiffness is accounted for. It is done through the displacement coupling correction coefficient ζ (see Equation (10)). It is also addressed by the stress modification terms in Equations (14) and (17).

High-speed permanent magnet motors operate under complex multiphysics coupling. The structural integrity of the rotor is crucial for the motor's overall reliability and service life. The introduction of axial constraints significantly alters the rotor's stress distribution. It also changes the associated failure mechanisms.

This section develops computational frameworks to evaluate three characteristic failure modes under axial constraints. These failure modes are interface debonding, permanent magnet fracture, and retaining sleeve yielding. These frameworks are based on the theory of elasticity, thermodynamics, and strength mechanics. They provide a theoretical basis for rational rotor strength design.

2.3.1. Evaluation Parameter Calculation for Interface Debonding Failure

Dynamic contact pressure serves as the principal evaluation parameter for interface debonding failure. This study presents an analytical model for this failure mechanism. The model accounts for the coupled effects of static interference, centrifugal displacement, and thermal displacement. A 3 kW permanent magnet motor is used to quantify the parameters in this computational framework.

Higher operational temperatures and rotational speeds cause relative movement at the contact interface. This movement arises from differences in the structural and material properties of the retaining sleeve and the permanent magnet. Accurately determining dynamic interference requires analysing the thermal and centrifugal displacement differences. The dynamic interference is calculated from the displacement responses of the retaining sleeve and permanent magnet under operating conditions. This calculation is defined by the governing equation (Equation (10)).

$$\begin{aligned} \delta_s &= b - c, \\ \Delta u_\omega &= u_{\omega i} - u_{\omega o}, \\ \Delta u_T &= u_{Ti} - u_{To}, \\ \delta_{eff} &= \delta_s - (\Delta u_\omega + \Delta u_T + \zeta \cdot \Delta u_\omega \cdot \Delta u_T) \end{aligned} \quad (10)$$

Here, b is the inner radius of the retaining sleeve, and c is the outer radius of the permanent magnet. δ_s is the static interference from the assembly design. $u_{\omega i}$ and $u_{\omega o}$ are the centrifugal displacements at the inner surface of the sleeve and the outer surface of the magnet, respectively. Δu_ω is the difference in centrifugal displacement across the interface. u_{Ti} and u_{To} are the thermal displacements at the same locations. Δu_T is the difference in thermal displacement. δ_{eff} represents the effective dynamic interference.

Here, ζ denotes the displacement coupling correction coefficient, ranging from 0.02 to 0.05. This coefficient accounts for the nonlinear coupling between centrifugal and thermal displacements, particularly when material properties, such as Poisson's ratio, differ between components. The value of ζ was determined through parametric finite element (FE) simulations and regression analysis, using experimental data from typical high-speed rotor materials (e.g., alloy 4169 and $\text{Sm}_2\text{Co}_{17}$ magnets). The coefficient increases as the difference in Poisson's ratio grows. A range of 0.02–0.05 was established to model the combined effect of effective dynamic interference accurately. A positive value of ζ indicates that the combined centrifugal and thermal expansion reduces the effective dynamic interference more than their linear sum does. This nonlinear reduction typically occurs when materials with different Poisson's ratios expand simultaneously under rotation and heating. The difference in Poisson's ratio is the key contributor to this coupled effect.

Based on the thick-walled cylinder theory [30], a radial stiffness coefficient K_r . It accounts for the temperature dependence of the material's elastic modulus (see Equation (11)). This approach extends the classical Lamé solution to include temperature effects [31]. For axially constrained rotors, the contact pressure model is proposed in Equation (12). This model enhances the classical interference-fit solution by incorporating dynamic effects [31,32]. The key dynamic parameters are the effective dynamic interference δ_{eff} and the radial stiffness coefficient K_r .

$$K_r = 1 + \frac{T}{500} \cdot \left(1 - \frac{E}{200} \right) \quad (11)$$

$$P = \frac{\delta_{\text{eff}} K_r}{b \left[\frac{1-\nu_1^2}{E_1} \left(\frac{a^2+b^2}{a^2-b^2} + \frac{\nu_1}{1-\nu_1} \right) + \frac{1+\nu_2}{E_2} \right]} \quad (12)$$

Here, T denotes the operational temperature. a is the outer radius of the retaining sleeve. E_1 and E_2 are the elastic moduli of the retaining sleeve and the permanent magnet, respectively. ν_1 and ν_2 are the corresponding Poisson's ratios. The interface debonding criterion is set as $P \geq 0$ to avoid vibrational and acoustic problems caused by microscopic gaps.

2.3.2. Evaluation Parameter for Permanent Magnet Fracture Failure

The permanent magnet is a brittle material. Its fracture behaviour is assessed using the maximum principal stress criterion, which is also referred to as the first strength theory. The corresponding failure condition is presented in Equation (13).

$$\sigma_{\text{max}} = \max\{|\sigma_r|, |\sigma_\theta|, |\sigma_z|\} \leq [\sigma_b] \quad (13)$$

Here, $[\sigma_b]$ denotes the maximum allowable tensile stress for the permanent magnet material.

Under axial constraint, the permanent magnet experiences a three-dimensional coupled stress state due to centrifugal loading. The analytical solutions for the centrifugal stress components are presented in Equation (14). These solutions are derived from the static equilibrium equations based on elasticity theory, with axial constraints taken into

account [29]. The base form of these equations corresponds to the classical solution for a rotating solid cylinder [33]. It is obtained when the $\eta\omega^2$ correction term is omitted.

$$\begin{aligned}\sigma_r &= \frac{3-2\nu_2}{8(1-\nu_2)}\rho_2\omega^2(c^2-r^2)(1+\eta\omega^2), \\ \sigma_\theta &= \frac{3-2\nu_2}{8(1-\nu_2)}\rho_2\omega^2\left(c^2-\frac{1+2\nu_2}{3-2\nu_2}r^2\right)(1+\eta\omega^2), \\ \sigma_z &= \nu_2(\sigma_r+\sigma_\theta)-\beta E_2\Delta T,\end{aligned}\quad (14)$$

Here, ω is the rotational angular velocity, ρ_2 is the density of the permanent magnet, ν_2 is its Poisson's ratio, and β is its coefficient of thermal expansion. The centrifugal stress correction coefficient η addresses stress concentration effects near the material's yield limit at ultra-high speeds. A small positive value, this coefficient captures the stress-redistribution-induced deviation from linear elasticity. This occurs as the material approaches its yield limit at ultra-high speeds due to localized plastic yielding or geometric discontinuities. This coefficient is calibrated through a comparison between the elastic analytical solution and elastoplastic finite element simulations. Its value depends on rotational speed, material yield behaviour, and geometric discontinuities.

Under axial constraint, Equation (14) indicates that the maximum centrifugal stress is located at the shaft centre ($r = 0$). The value of this maximum stress is expressed by Equation (15).

$$\sigma_{\omega\max} = \frac{3-2\nu_2}{8(1-\nu_2)}\rho_2\omega^2c^2(1+\eta\omega^2) \quad (15)$$

The maximum principal stress $\sigma_{\max}$ in the constrained permanent magnet is the sum of the centrifugal stress at the rotor centre and the stress induced by the interference fit. It is expressed in Equation (16).

$$\sigma_{\max} = \sigma_{\omega\max} - \xi \cdot p \quad (16)$$

Here, ξ (typically 1.0–1.3) corrects for the local stress concentration at the magnet-sleeve interface due to the interference fit. It converts the nominal contact pressure p into an equivalent tensile stress amplification factor at the magnet's critical point. The corresponding fracture criterion is defined as $\sigma_{\max} \leq 0.8\sigma_b$. This limit incorporates a safety factor of 1.25. It restricts the allowable stress to 80% of the ultimate tensile strength σ_b . This safety margin accounts for the brittleness of sintered $\text{Sm}_2\text{Co}_{17}$ magnets, potential dynamic loads, and stress concentrations that are not fully modelled in the static analysis. It also follows established design practices for brittle materials under high-speed cyclic loading [34,35].

2.3.3. Evaluation Parameter for Sleeve Yield Failure

Sleeves are typically fabricated from ductile metals. Their yield failure is assessed using the von Mises stress criterion, also known as the fourth strength theory. The evaluation model is given by Equation (17). This model enhances accuracy under combined loads by incorporating axial stress coupling and thermal correction terms. The term inside the square root corresponds to the standard von Mises stress formula for cylindrical coordinates [29]. It is multiplied by a speed-dependent correction factor $(1 + \psi \cdot \omega)$.

$$\sigma_v = \sqrt{\frac{1}{2}[(\sigma_r - \sigma_\theta)^2 + (\sigma_\theta - \sigma_z)^2 + (\sigma_z - \sigma_r)^2]} \cdot (1 + \psi \cdot \omega) \quad (17)$$

Here, ψ is the rotational speed correction coefficient. We determined its value through regression analysis of parametric finite element simulations. The coefficient ψ quantifies the stress-distribution deviations that arise under dominant centrifugal loads, where the

uniform axial stress assumption no longer holds. A positive ψ signifies an increase in equivalent stress caused by dynamic stiffening and stress-path alterations during high-speed rotation. The yield criterion is given by $\sigma_v \leq [\sigma]$, with the allowable stress $[\sigma] = 900$ MPa. This value corresponds to a safety factor of approximately 1.15 relative to the material yield strength of 1030 MPa. A safety factor of approximately 1.15 is applied. This value is derived from two considerations: a factor of 1.1 for high-cycle fatigue strength reduction under combined stresses at 200,000 r/min, and an additional allowance of 1.05. This allowance covers stress concentrations arising from manufacturing tolerances. For example, surface roughness or micro-geometric imperfections [36].

Stress assessment is performed exclusively at the inner surface of the retaining sleeve ($r = b$). We identify this location as the critical point where the maximum stress consistently occurs.

2.4. Finite Element Analysis Methodology

This section describes the finite element analysis methodology used to validate the analytical model and assess rotor strength failure modes.

Three-dimensional stress simulations of the retaining sleeve and permanent magnets were performed using ANSYS Workbench. The interference fit was modeled by applying prescribed interference values at the magnet-sleeve interface. Axial constraints were applied by fixing the axial displacement ($U_z = 0$) at both ends of the thrust disk and lock nut. This simulates the constraint from end plates and bolts. Contact interfaces between the sleeve and magnet, as well as between the magnet and shaft, were defined as bonded contacts. Friction coefficients of 0.15 and 0.1 were assigned, respectively, allowing stress transfer while preventing sliding.

The rotor assembly was discretised with a structured mesh (Figure 3). The mesh is refined in critical interfacial areas to strike a balance between computational accuracy and efficiency.

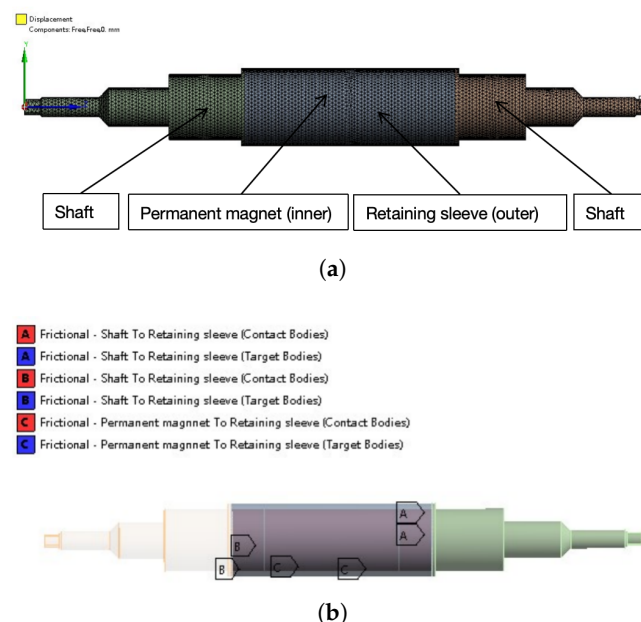


Figure 3. Finite element model of the rotor assembly subject to axial constraints: (a) Mesh Generation Diagram of the Shafting System. (b) Constraint Conditions of the Shafting Contact Surfaces. The model was developed in ANSYS Workbench employing a structured tetrahedral mesh with a specified element size of 1 mm.

3. Results and Discussion

This study validates the proposed methodology through an analysis of stresses associated with three rotor failure modes. The investigation focuses on a 3 kW, 200,000 r/min solid-rotor high-speed permanent magnet motor designed for air compressor applications. We validated the model by integrating analytical modelling, finite-element simulation, and experimental verification.

3.1. Stress Calculation and Analysis of Solid Rotor Failure Modes

3.1.1. Core Evaluation Parameter for Interface Debonding Failure

We calculated the contact pressure within a 3 kW permanent magnet motor under axial constraint using the developed interface debonding parameter model. The calculations (Table 3) covered specified ranges of rotational speed and interference fit.

Table 3. Interface Stress Calculation Results.

Interference (mm)	Maximum Compressive Stress at Interface (MPa)				
	0 r/min	50,000 r/min	100,000 r/min	150,000 r/min	200,000 r/min
0.04	69.527	68.247	64.638	58.410	48.841
0.06	104.310	102.980	99.395	93.293	84.452
0.08	139.720	138.250	134.420	128.080	119.400
0.10	175.900	174.330	170.370	163.760	154.620
0.12	212.040	210.440	206.420	199.770	190.430

An analysis of Table 3 shows that contact pressure decreases nonlinearly with rotational speed. This trend holds for all interference-fit values examined. For example, with a 0.08 mm interference fit, the pressure drops from 139.72 MPa at 0 r/min to 119.4 MPa at 200,000 r/min. It represents a reduction of 14.5%. The decrease results from centrifugal force, which induces radial displacement between the retaining sleeve and the permanent magnet. This displacement reduces the effective interference fit, thereby lowering the contact pressure. At higher speeds, the centrifugal effect intensifies. It leads to a greater attenuation of compressive stress.

Contact pressure shows a strong positive correlation with interference magnitude at a fixed rotational speed. Under static conditions (0 r/min), an increase in interference from 0.04 mm to 0.12 mm raises the contact pressure from 69.527 MPa to 212.04 MPa. This change represents an increase of 205%. It confirms that larger interference fits produce higher initial contact pressure during assembly. This higher initial pressure helps counteract centrifugal relaxation during high-speed operation. Contact pressure decreases with speed for all interference values. Configurations with higher initial interference retain a greater proportion of their initial pressure.

The interfacial stress distribution (see Section 3.2 for details) supports this finding. It shows that the contact pressure gradually decreases with increasing speed. A comparison reveals a significant reduction in compressive stress concentration at the magnet-sleeve interface. This reduction occurs at 200,000 r/min compared to static conditions. This observation confirms the quantitative trends in Table 3. The evolution of the stress contours illustrates how centrifugal loading redistributes interfacial pressure. The most excellent attenuation occurs in the central region of engagement.

In summary, the interference fit governs the baseline contact pressure. The rotational speed determines the rate of pressure decrease. Since the rotational speed dictates the rate of pressure decrease, the interference value must be selected according to the system's maximum operating speed. This design criterion ensures that sufficient assembly stress is

achieved. It also maintains safe interfacial pressure at high speeds. This dual action prevents interface debonding. Debonding can occur if contact pressure becomes insufficient.

3.1.2. Evaluation Parameter for Magnet Fracture Failure

We calculated the maximum principal stress for a 3 kW axially constrained motor using a developed fracture evaluation model. The calculation spanned a range of rotational speeds and interference fits. In this analysis, compressive and tensile stresses are defined as negative and positive, respectively (Table 4).

Table 4. Maximum Stress Calculation Results for Permanent Magnet.

Interference (mm)	Maximum Tensile Stress in Magnet (MPa)				
	0 r/min	50,000 r/min	100,000 r/min	150,000 r/min	200,000 r/min
0.04	−43.319	−36.381	−15.602	21.452	84.793
0.06	−64.986	−58.045	−37.254	−2.694	53.688
0.08	−85.792	−78.966	−58.492	−24.309	26.365
0.10	−105.100	−98.351	−78.112	−44.385	3.066
0.12	−124.260	−117.530	−98.333	−63.681	−16.579

The stress state of the permanent magnet shifts as speed increases at a constant interference level (Table 4). The shift is from compressive to tensile stress. In the case of 0.08 mm interference, the stress changes from −85.792 MPa at 0 r/min. At 200,000 r/min, it becomes +26.365 MPa. It is a total shift of 110 MPa. The trend confirms that centrifugal effects intensify with speed. They overcome the compressive stress from the interference fit. This critical transition from compressive to tensile stress significantly increases the risk of brittle fracture.

At a fixed rotational speed, the interference fit determines the stress state of the magnet. Larger interference values create a larger compressive stress reserve. This reserve delays the transition from compressive to tensile stress. Consider the condition at 200,000 r/min. With a 0.04 mm interference fit, the magnet experiences a tensile stress of 84.793 MPa. With a 0.12 mm interference fit, it retains a compressive stress of −16.579 MPa. These results indicate that selecting an optimal interference fit can improve mechanical performance at high speeds. This choice increases assembly complexity. It also raises stress levels in the retaining sleeve.

Figure 5 clearly shows the transition from a compressive to a tensile stress state. We simulated the tensile stress distribution under operational conditions. This concentration occurs at the shaft centre ($r = 0$), consistent with the theoretical prediction derived from Equation (15). The model successfully identifies stress localisation patterns. These patterns are critical for assessing fracture risk.

3.1.3. Calculation of Evaluation Parameters for Sleeve Yield Failure

We developed a parametric model to assess the yield failure of retaining sleeves and performed systematic calculations using it. The goal was to determine the von Mises stress distribution. The distribution is within the retaining sleeve of a 3 kW permanent magnet motor. The analysis examined the effects of rotational speed and interference-fit magnitude under axially constrained boundary conditions. All numerical results are presented in Table 5.

The von Mises stress in the retaining sleeve increases steadily with rotational speed for all interference values (Table 5). Stress increases consistently from 0 r/min to 200,000 r/min. For a 0.06 mm interference fit, the equivalent stress increases from 792 MPa to 814 MPa, representing a relative increase of approximately 3%. Centrifugal loading is the leading

cause of this stress increase. The absolute increase is modest. However, it is not negligible at ultra-high speeds as the total stress nears the material's allowable limit. Therefore, this effect requires careful attention during structural design.

Table 5. Von Mises Stress Calculation Results for Sleeve.

Interference (mm)	Von Mises Stress in Sleeve (MPa)				
	0 r/min	50,000 r/min	100,000 r/min	150,000 r/min	200,000 r/min
0.04	528.53	529.81	533.83	540.59	554.52
0.06	792.78	794.12	798.30	804.77	814.99
0.08	1057.00	1058.20	1062.10	1069.20	1078.60
0.10	1321.10	1322.40	1326.30	1333.70	1342.80
0.12	1585.30	1586.50	1590.50	1596.80	1607.20

The interference fit is the main factor controlling stress in the retaining sleeve (Table 5). At any fixed speed, increasing interference by 0.02 mm raises stress by about 264–265 MPa. Under static conditions (0 r/min), interference was increased from 0.04 mm to 0.12 mm. It raised the von Mises stress from 528.53 MPa to 1585.3 MPa. The increase is threefold. The von Mises yield criterion is used to assess sleeve failure. It requires $\sigma_{vm} \leq 900$ MPa. This requirement sets the allowable interference limit at $\delta \leq 0.06$ mm. The design must consider both sleeve integrity and interface clamping needs. The 0.06 mm interference fit is identified as the optimal design.

Figure 5 shows the von Mises stress distribution. The stress is highest at the inner surface of the retaining sleeve, confirming this as the critical region. The observed pattern supports a theoretical approach.

3.2. Stress Calculation and Comparison with Finite Element Analysis

Using the finite element methodology described in Section 2.4, stress distributions in the permanent magnets, sleeve, and contact interface under axial constraint were calculated (see Figures 4 and 5). The mathematical solutions for both constraint conditions agree with the results of the finite element analysis. This agreement validates the proposed computational framework and the evaluation parameters used to assess multiple rotor strength failure modes (Tables 3–5).

Table 6 provides a clear comparison, including both analytical and finite element (FE) solutions evaluated under rated operating conditions for three failure modes (200,000 r/min, $\delta = 0.06$ mm). Excellent agreement is observed with relative errors below 1%, confirming the high accuracy of the analytical model.

Table 6. Comparison of analytical and finite element results for three failure modes under rated conditions.

Failure Mode/Parameter	Analytical Result	Finite Element Result	Relative Error
Interface Pressure (MPa)	84.45	84.99	0.64%
Max. Tensile Stress in Magnet (MPa)	53.69	54.12	0.80%
Von Mises Stress in Sleeve (MPa)	814.99	815.50	0.06%

Figure 4 presents the finite element analysis (FEA) results under rated conditions (200,000 r/min, $\delta = 0.06$ mm). Figure 4a shows the contact pressure distribution at the sleeve-magnet interface. The uniform compressive stress (blue-green) indicates that the effective interference is maintained without localised separation. Figure 4b illustrates the tensile stress in the permanent magnet. The highest stress (red region) occurs at the rotor centre ($r = 0$), consistent with Equation (15). This confirms the critical area for fracture

risk. Figure 4c displays stress on the sleeve's inner surface. Elevated stress levels (yellow to red) are observed at $r = b$, matching the theoretical prediction that identifies the most critical location for yield assessment.

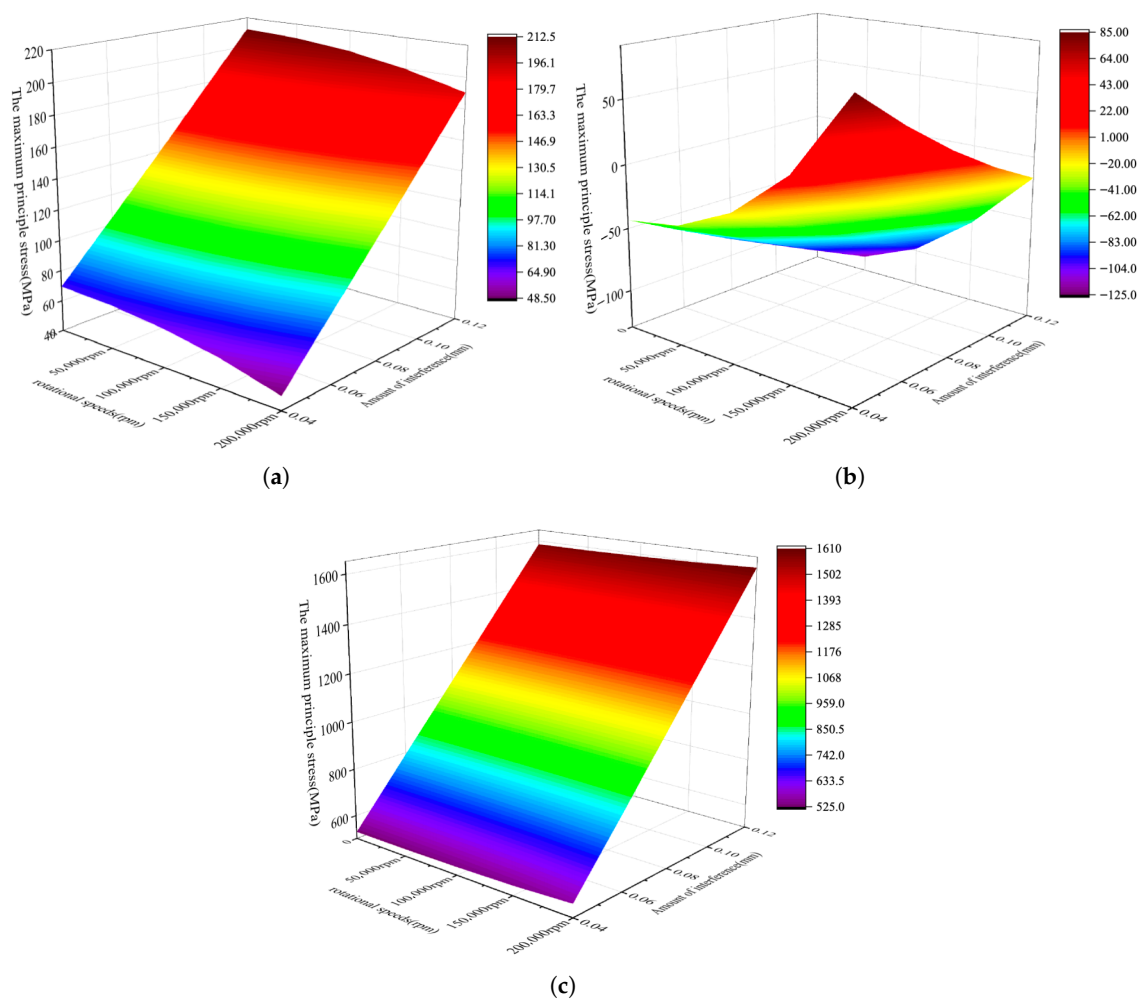


Figure 4. Rotor stress distributions under rated operating conditions computed using the finite element method: (a) Contact surface stress distribution. (b) Permanent magnet stress field. (c) Inner surface stress of the sleeve.

These distribution characteristics show excellent agreement with the quantitative results in Table 6. All discrepancies remaining below 1%. It verifies the accuracy of the computational framework presented in this work.

Figure 5 shows the spatial distribution of internal stress fields within the rotor. Figure 5a displays the maximum von Mises stress at the sleeve inner edge, with a value of 841.99 MPa. It corresponds to the stress under a 0.06 mm interference fit at 200,000 r/min (Table 5). The corresponding safety factor is 1.22 (Equation (18)). Figure 5b presents the tensile stress field in the permanent magnet. The stress reaches its maximum at the shaft centre, measuring 53.688 MPa. This result is consistent with the stress state for a 0.06 mm interference (Table 4). Figure 5c displays the distribution of interfacial pressure. The distribution is uniform, with the maximum pressure located at the outer edge of the magnet (84.452 MPa). This trend matches the data in Table 3. For clarity, Figure 4 visually connects the analytical results (Tables 3–5) with the actual stress patterns. This connection is explained as follows:

- Figure 4a shows the contact pressure at the interface. This result relates to the interface debonding analysis in Table 3. The figure confirms that compressive stress is maintained.
- Figure 4b illustrates the tensile stress in the permanent magnet. This visualisation supports the fracture risk prediction from Table 4.
- Figure 4c displays the von Mises stress on the inner surface of the sleeve. It confirms the yield assessment findings in Table 5.

In summary, the stress distributions obtained from finite element analysis (presented in Section 3.2) closely match the analytical values. The errors are all within 1%, as shown in Table 6. This close agreement validates the proposed model comprehensively.

$$SF_{\text{sleeve}} = \frac{\sigma_{\text{yield}}}{\sigma_{\text{vm,max}}} \quad (18)$$

Here, σ_{yield} is the material yield strength and $\sigma_{\text{vm,max}} = 841.99$ MPa. Stress is highest near the shaft interfaces. The central region, adjacent to the permanent magnets, has significantly lower stress. The outer surface shows almost no stress concentration.

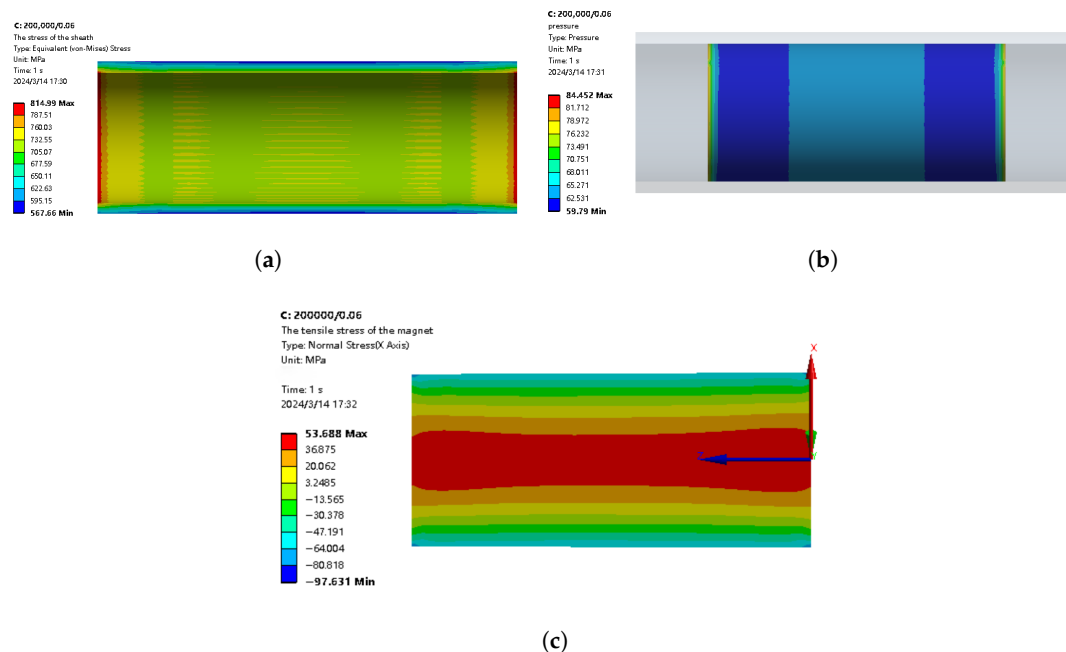


Figure 5. Spatial distribution of stress fields within the rotor assembly: (a) Von Mises equivalent stress in the sleeve. (b) Compressive stress distribution at the contact interface. (c) Tensile stress field in the permanent magnets under combined centrifugal load and axial constraint.

The permanent magnet experiences a maximum radial tensile stress of 53.688 MPa. It results from centrifugal loading combined with axial constraint effects. They alter the radial and circumferential stress distributions. Equation (19) gives a corresponding safety factor of 1.58.

$$SF_{\text{magnet}} = \frac{\sigma_{\text{ult}}}{\sigma_{\text{rad,max}}} \quad (19)$$

Here, $\sigma_{\text{rad,max}} = 53.688$ MPa and σ_{ult} is the ultimate tensile strength. The stress decreases steadily from the central region outward, reaching its lowest value at the outer surface. The contact pressure at the interface reaches a maximum of 84.452 MPa at the outer edge of the magnet. Equation (20) gives a safety factor of 7.10.

$$SF_{\text{contact}} = \frac{p_{\text{crit}}}{p_{\text{max}}} \quad (20)$$

Here, $p_{\max} = 84.452$ MPa. The pressure distribution is mostly uniform. It decreases gradually from the central region toward the edges.

A comparison reveals that the predicted failure locations and evaluation parameters closely align. The evaluation parameters also align with the numerical results. It confirms the validity of the developed analytical framework.

3.3. Comparison with Prior Studies

We compare our results with key published studies focusing on interference-fit rotor mechanics.

(1) Centrifugal relaxation of interface pressure

This study observes a nonlinear decrease in contact pressure with increasing speed (Table 3). It confirms the relaxation effect in rotating interference-fit assemblies [5,22]. For example, the trend aligns with the results of Yang et al. [22], who attributed pressure decay primarily to differential radial expansion. Their 2D plane-stress model does not include axial boundary constraints. The current 3D model explicitly consists of this effect, providing a more complete description. The model predicts that approximately 81% of the static pressure is retained at 200,000 r/min for a 0.06 mm fit. It aligns with the expected performance range for high-strength steel retaining sleeves [6].

(2) Tensile stress onset in permanent magnets

A stress transition occurs in the permanent magnet. It changes from compressive to tensile. It happens at a critical speed (Table 4). The limit is for high-speed SPM rotors [7,8]. This study observes that the critical speed depends on the interference fit. Li et al. [20] reported these trends, focusing on interior permanent magnet synchronous machines. However, a direct quantitative comparison is limited. The limitation arises from differences in rotor topology. One uses surface-mounted magnets. The other uses interior magnets. The present analytical model provides specific insights for axially constrained solid rotors. This configuration exhibits distinct behaviour in stress transition. Li et al. [20] did not separately examine this behaviour.

(3) Sleeve stress and interference design limit

The von Mises stress exhibits a near-linear increase. This stress is within the retaining sleeve. The increase is attributed to the interference magnitude (Table 5). The principles are mechanical [5,17]. For the studied configuration, the derived safe operational limit for interference is $\delta \leq 0.06$ mm. We applied appropriate scaling accounts for differences. The differences are in three aspects: rotational speed, geometry, and material properties. Stress magnitudes were then predicted. The predictions show satisfactory agreement. The agreement is with published finite element analysis results [17]. These results pertain to a composite rotor operating at 28,000 r/min.

In conclusion, this study addresses three failure mechanisms in high-speed rotors. They also include permanent magnet tensile stress. They include interface relaxation, magnet tensile stress, and sleeve yielding. Previous analyses have relied mainly on finite element simulations or two-dimensional analytical models. The principal contribution of this work is a three-dimensional analytical framework. It explicitly quantifies the influence of axial constraints. This framework offers a more precise analytical tool. It is used for defining safe operating boundaries in the design of high-speed rotors.

3.4. Experimental Validation

Direct measurement of internal stresses in rotor components is technically challenging at ultra-high speeds up to 200,000 r/min. Due to harsh operating conditions, it includes

sensor installation difficulties, signal transmission limitations, high temperatures, and significant centrifugal forces. Therefore, this study adopts an indirect validation strategy based on the rotor's dynamic response. Stress redistribution and amplification under axial constraint (Sections 3.1 and 3.2) induce micro-deformations in the rotor structure. These micro-deformations alter the dynamic characteristics of the rotor-bearing system, ultimately manifesting as measurable changes in the shaft orbit [37].

To capture this dynamic response and correlate it with model predictions, a dedicated experimental platform was constructed (Figure 6). The key instrumentation consists of two high-precision eddy-current displacement sensors. As illustrated in Figure 6b, these non-contact sensors are mounted orthogonally at the compressor end. They measure radial displacement in the horizontal (X) and vertical (Y) directions, enabling real-time reconstruction of the shaft orbit at that plane. This provides direct insight into the lateral dynamic motion state of the rotor.

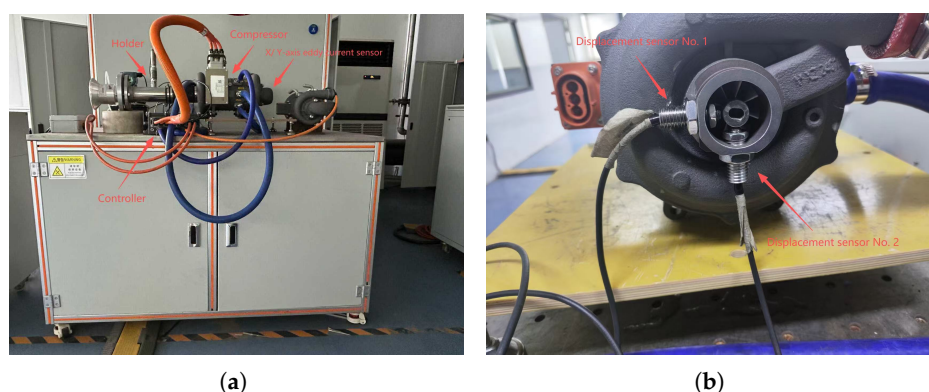


Figure 6. Schematic representation of the experimental prototype test platform: (a) Schematic diagram of the experimental test platform and sensor configuration. (b) Enlarged view showing the orthogonal arrangement of two eddy-current displacement sensors (Model ES-S1M8-CA3.0mB0, indicated by red circles and labelled Sensor 1 and Sensor 2) near the compressor end of the shaft. These sensors provide real-time, non-contact measurements of radial displacement. The data are used to reconstruct the shaft orbit. It indirectly infers stress-induced micro-deformations for model validation.

The model quantifies the stress amplification caused by axial constraints as speed increases, as seen in the rising von Mises stress of the sheath. This predicted change leads to a corresponding increase in shaft orbit amplitude, a parameter directly measured by the sensors. Comparing the experimental increase in orbit amplitude (from 180,000 to 200,000 r/min) with the modelled stress trend. It provides strong indirect validation of the analytical framework. To isolate the contribution of stress-induced stiffness variation, a comprehensive characterization of the system's dynamic properties was conducted before the main experiment. This step ensured that rotor-dynamic resonance was not the cause of the observed amplitude change.

The system's dynamic characteristics were fully characterized. It ensured that the observed vibration changes were attributed to stress-induced stiffness variation, not to rotor-dynamic resonance. The first-order bending critical speed ($\omega_c = 272,380$ r/min) is well separated from the operating range (180,000–200,000 r/min), as shown in the Campbell diagram (Figure 7). The calculated safety margin ($\eta_m = (\omega_c - \omega_{\max}) / \omega_c \approx 0.266$) substantially exceeds the standard design threshold of 0.20 [38]. This quantitative finding rigorously excludes excitation of the bending critical mode during testing. Consequently, any vibration changes within the operating range can be confidently attributed to non-resonant drivers, primarily the stress-state change predicted by the model.

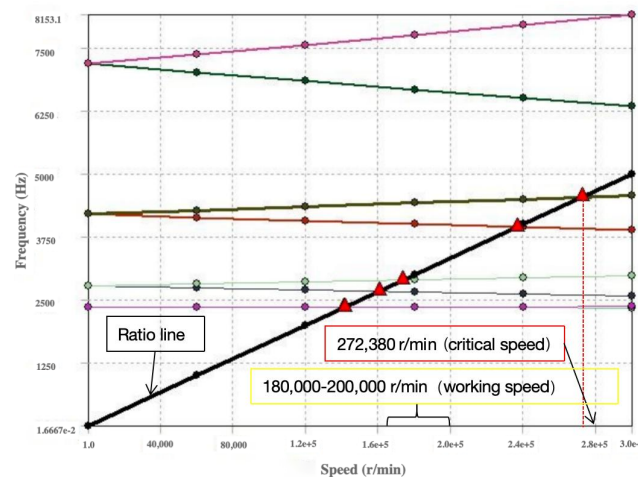


Figure 7. Campbell diagram of the rotor system. The area enclosed by the X-axis bracket represents the operational speed range (180,000–200,000 r/min). The critical speed is identified at 272,380 r/min, yielding a safety margin of $\eta_m \approx 0.266$ relative to the maximum operational speed. The variation in the first-order bending natural frequency with rotational speed is plotted as a black solid line. A red pentagram explicitly marks the critical speed.

$$\eta_m = \frac{\omega_c - \omega_{\max}}{\omega_c} \quad (21)$$

Here, $\omega_c = 272,380$ r/min is the critical speed, and $\omega_{\max} = 200,000$ r/min is the maximum operating speed. The resulting value of 0.266 exceeds the 0.2 design threshold, meeting rotor-dynamics requirements. This outcome confirms the structural stability of the axially constrained rotor. It further supplies a theoretical foundation for the shaft-orbit experiment shown in Figure 8.

The amplitude of the shaft orbit R is defined as the maximum radial displacement from its centre. For each data point, the instantaneous radius is computed as $R_{\text{inst}} = \sqrt{X^2 + Y^2}$, where X and Y are the orthogonal displacement signals. The reported amplitude R_{max} corresponds to the maximum value of R_{inst} during steady-state operation. Based on sensor data (Figure 8), R_{max} increased from 55 μm at 180,000 r/min to 65 μm at 200,000 r/min. This represents a calculated increase of approximately 18%.

This experimental trend confirms the stress amplification predicted by the model under axial confinement. While the model indicates only a +2.8% increase in sheath von Mises stress (within the same speed range, with $\delta = 0.06$ mm), the observed 18% vibration amplification can be explained by rotor dynamics. It is the rising stress, concentrated on the sheath's inner surface, that enhances the local structural stiffness. This altered stiffness modifies the system's response to residual imbalance, leading to amplified vibration. Such an effect, where static stress influences dynamic response via stiffness adjustment, is a recognized behaviour of rotor-bearing systems.

The increase in shaft orbit amplitude provides strong indirect evidence that the stress field amplifies as predicted. Furthermore, the model's validity is supported by operational stability under the selected design ($\delta = 0.06$ mm). It adheres to the sheath yield criterion ($\sigma_v \leq 900$ MPa). The closed and stable orbit in Figure 8 indicates no vibration instability. The absence of interface-separation peaks in the vibration spectrum is consistent with the model's prediction of positive contact pressure (Table 3). This stability is confirmed by two observations.

Future work may consider strain gauges or fibre Bragg grating sensors. These sensors enable direct stress measurement at critical rotor locations under low-speed or static condi-

tions. It helps to establish a quantitative link between stress and vibration response. This approach will provide a more direct basis for experimental validation.

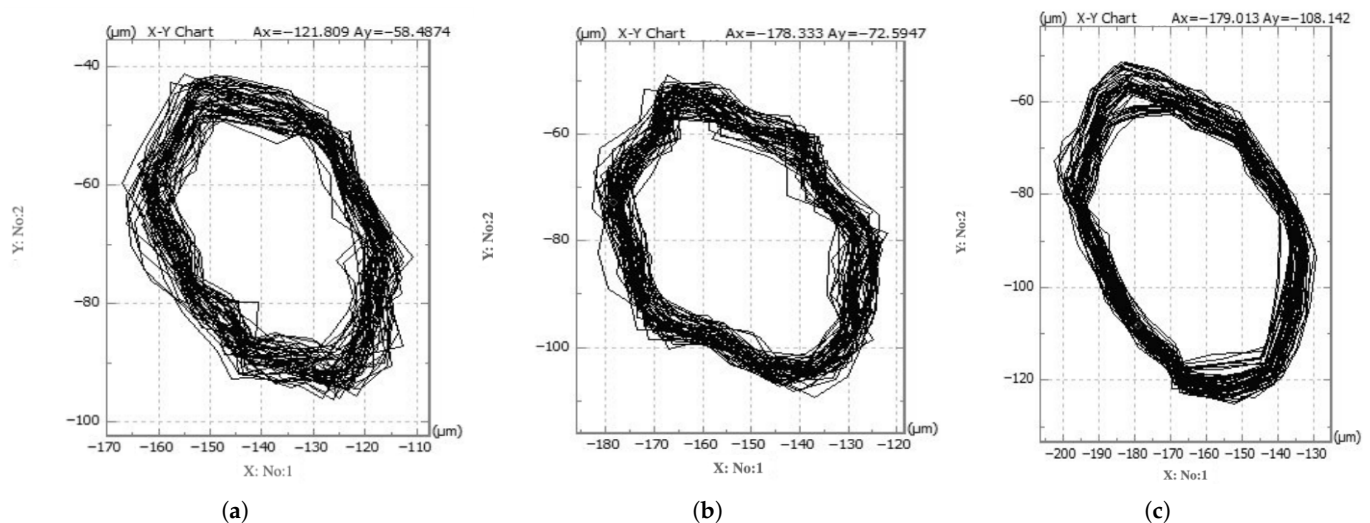


Figure 8. Central trajectory diagram: (a) $n = 180,000$ r/min. (b) $n = 190,000$ r/min. (c) $n = 200,000$ r/min. The abscissa represents the first displacement sensor, and the ordinate represents the second displacement sensor.

4. Discussion

4.1. Model Limitations and Applicability

The present analytical framework is based on two main assumptions: complete material isotropy and uniform temperature distribution. These simplifications maintain sufficient accuracy during initial design stages. The model has limitations. It would require enhancement for specific rotors. These rotors may employ anisotropic composite sleeves. They may also experience substantial radial temperature gradients. The static formulation accurately describes steady-state stress patterns. However, it lacks dynamic contributions from mass imbalance. They also arise from associated vibration responses.

Furthermore, empirical correction coefficients (ζ, ψ, η) were introduced into the evaluation models to improve prediction accuracy for specific high-speed coupling effects. These coefficients were calibrated using finite element simulation results. The current model focuses on the mechanical effects of axial constraints and does not consider electromagnetic forces from motor operations, such as Lorentz forces and magnetic pressure. Such forces could introduce additional stress components, particularly under loaded or transient conditions. For the present steady-state, no-load mechanical stress analysis, this omission is justified.

Experimental validation at ultra-high speeds presents significant complexity. Consequently, this study primarily relies on indirect vibration analysis. Future work will incorporate direct stress measurements. Embedded strain sensors enable this monitoring/validation under low-speed or simulated loading conditions. Such measurements will provide more direct evidence for validating the analytical model.

4.2. Quantitative Correlation Between Experimental and Simulation Results

This study validates the model through indirect vibration measurements. The quantitative data clearly link model predictions with experimental observations (Figure 7). This correlation is evident in three key aspects.

1. **Trend Consistency:** The model predicts a +2.8% increase in sleeve stress. Experimentally, the vibration amplitude increases by +18%. Both values move in the same

direction. It matches a rotor dynamics principle: when stress rises, stiffness changes, amplifying the vibration response.

2. Interface-stability verification: No frequency peaks indicating interface separation are found in the vibration spectrum. It matches the model's prediction. The predicted contact pressure stays positive (84.45 MPa) at rated speed. Therefore, the interface is likely intact during operation.
3. Dynamic verification of the stress-state transition: According to Table 4, the permanent magnet transitions from compressive to tensile stress. This change does not cause vibration instability, as shown by the closed orbits in Figure 8. The result indicates that the stress-state transition is dynamically acceptable. It also agrees with the static-strength prediction of the model.

These multi-parameter, quantitative correspondences provide strong support for the model's predictive capability under ultra-high-speed operating conditions. Direct stress measurement at such speeds is currently impractical due to technical constraints. However, the indirect validation framework is based on well-established physical mechanisms, which ensure its high scientific credibility.

4.3. Practical Implementation Guidelines

An interference limit of 0.06 mm is established, providing a reliable design reference for motors with similar ratings. In applications subject to repeated start–stop cycles or variable loading, larger safety margins are recommended. These enhanced margins contribute to ensuring long-term structural integrity. The validated computational strategy enables designers to balance two key requirements: the clamping needs at the interface and the stress limits within the sleeve. This balance is especially critical when advanced high-strength sleeve materials are employed.

This research provides a methodological foundation for rotor stress analysis under axial constraints. A finite element model (Figure 3) serves as the core of this foundation. The model enables detailed computational analysis of the constrained stress state. Quantitative model validation is performed by correlating simulated stress fields (Figures 4 and 5) with analytical results (Tables 3–6). This comparison confirms the predictive accuracy of the proposed assessment models. The models address three primary failure modes: interface debonding, magnet fracture, and sleeve yielding.

Operational safety is also assessed from a rotor dynamics perspective. The Campbell diagram (Figure 7) is used for this assessment. The diagram shows adequate separation between the first bending critical speed and the operational speed range. This result validates the dynamic integrity of the system. Experimental support is obtained from shaft orbit measurements (Figure 8). An increase in orbit amplitude is observed as the speed rises. This growth indirectly confirms the predicted stress amplification resulting from axial constraint.

Collectively, these analytical, numerical, and experimental results constitute a coherent verification framework. This framework operates as a closed-loop system. The framework systematically explains how axial constraints govern failure in high-speed permanent magnet rotors. Results show that constraint-induced stress redistribution has two main effects. It increases static stress levels and amplifies the dynamic response. These effects define the operational limits of the rotor system.

5. Conclusions

This study presents a static analytical model for axially constrained high-speed permanent magnet rotors. The influence of rotational speed and interference fit on three key failure mechanisms was systematically investigated to improve evaluation methods.

Integration with finite element simulations and experimental validation was achieved using a 3 kW prototype. This integrated approach yielded the following main conclusions:

1. A static analytical model was developed to predict the mechanical behaviour of high-speed rotors under axial constraints. Based on three-dimensional axisymmetric elasticity theory, this model calculates the stress field. It explicitly accounts for the influence of axial constraint. It thereby provides engineers with a practical and reliable tool. This tool quantifies constraint effects. It also accurately assesses key performance indicators. Consequently, the model supports failure risk evaluation. This includes risks such as retaining sleeve yielding and permanent magnet fracture.
2. Refined models are proposed for the three primary failure modes: interface debonding, permanent magnet fracture, and retaining sleeve yielding. These models parameterise critical effects such as dynamic interference fit. They provide a validated design tool that improves the predictive accuracy and operational reliability of high-speed rotor systems.
3. The systematic study clarifies the separate roles of rotational speed and interference fit. Interference fit sets the baseline interface contact pressure. Rotational speed mainly governs how this pressure decays under rotation. As a result, the stress state in the permanent magnet is composed of compressive stress from the interference fit and tensile stress from centrifugal loading. On the other hand, the von Mises stress in the sleeve, which governs yield risk, rises steadily with rotational speed.
4. The proposed model is validated through quantitative correlation between finite-element simulations and experimental responses. The model's predicted stress trends closely match the measured vibration responses in numerical values (Table 7). In addition, the vibration spectrum contains no peaks that would indicate interface separation. This observation agrees with the model's prediction that contact pressure remains positive. These indirect, yet quantitative, validations, based on multiple physical indicators, provide reliable support for the model's accuracy under ultra-high-speed conditions.

Table 7. Quantitative correlation between model predictions and experimental observations for the optimum design ($\delta = 0.06$ mm).

Predicted Quantity (Source)	Predicted Change (200k r/min)	Experimental Metric	Observed Change (180k $\rightarrow$ 200k r/min)	Physical Correlation & Conclusion
Sleeve Von Mises Stress (Table 5, $\delta = 0.06$ mm)	792.78 $\rightarrow$ 814.99 MPa (+2.8%)	Orbit amplitude (Figure 8)	+18% (amplitude growth)	Stress increase leads to rotor stiffening and vibration amplification. Trend consistency supports the model.
Interface Contact Pressure (Table 3, $\delta = 0.06$ mm)	104.31 $\rightarrow$ 84.45 MPa (−19.0%)	Frequency spectrum	No separation-related peaks	Positive pressure maintained. No instability observed, validating interface integrity.
Magnet Tensile Stress (Table 4, $\delta = 0.06$ mm)	−64.99 $\rightarrow$ +53.69 MPa (Comp. to Tensile)	Orbit shape/stability (Figure 8)	Orbits remain closed and stable	Stress state transition does not induce instability, confirming design acceptability.

Note: Experimental amplitude increase is calculated from orbit radii in Figure 8. The vibration amplification factor (≈ 6.4) reflects the mechanical gain between the stress change and the dynamic response.

Future research will focus on: (i) Extending the current mechanical model with high-fidelity thermal analysis. CFD and thermal network analysis enable such analysis. The goal is to account for non-uniform temperature fields resulting from high-speed losses. (ii) Analysing the failure behaviour of advanced retaining-sleeve materials, especially carbon-fibre composites, under axially constrained configurations. (iii) Developing condition-monitoring systems for rotor-health assessment and service-life prediction.

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Abbreviations

The following abbreviations are used in this manuscript:

HSPMMs	High-Speed Permanent Magnet Machines
FE	Finite Element
SPM	Surface-Mounted Permanent Magnet
UAV	Unmanned Aerial Vehicle
FEM	Finite Element Method

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